Report 2523

NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Washington, D.C. 20007

AD659053

THE DTMB PLANAR-MOTION-MECHANISM SYSTEM

by

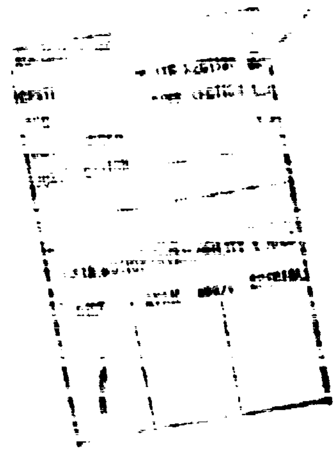
Morton Gertler

The material in this report was originally presented as a paper before the Symposium on the Towing Tank Facilities, Instrumentation, and Measuring Techniques held in Zagreb, Yugoslavia from 22 to 25 September 1959 on the occasion of the official opening of the Shipbuilding Research Institute.

HYDROMECHANICS LABORATORY
TEST AND EVALUATION REPORT

July 1967 Report 2523

THE DTMB PLANAR-MOTION-MECHANISM SYSTEM



The Naval Ship Research and Development Center is a U.S. Navy center for laboratory effort directed at achieving improved sea and air vehicles. It was formed in March 1967 by merging the David Taylor Model Basin at Carderock, Maryland and the Marine Engineering Laboratory at Annapolis, Maryland.

Naval Ship Research and Development Center
Washington, D.C. 20007

PRECEDING PAGE BLANK

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

SYNOPSIS

This paper describes and discusses the underlying principles of a planar-motion-mechanism system which was designed, constructed, and recently put into service at the David Taylor Model Basin. The DTMB Planar-Motion-Mechanism System incorporates in one device a means for experimentally determining all of the hydrodynamic-stability coefficients required in the equations of motion for a submerged body in six degrees of freedom. These include coefficients usually classified into the three categories of static-stability, rotary-stability, and acceleration derivatives.

The most unique feature of the system is the method used to impart hydrodynamically pure pitching and heaving motions to a given submerged body. This enables the explicit and accurate determination of individual derivatives without resort to the solution of simultaneous equations as is necessary when other types of oscillation devices are used. Other combinations of pitching and heaving motions can also be produced by the mechanism, if so desired.

The balance system used to measure the forces and moments also differs distinctly from the multi-component dynamometers used by most other model-basin or wind-tunnel facilities. It is composed of modular flexural gages employing a variable-reluctance transducer which individually measure a single force in either the X-, Y-, or Z-direction depending upon orientation. Roll moment is obtained by a torsional gage which is sensitive only to a moment about a single axis. A balance system is thus produced which is mechanically free of interactions and consequently the calibration of each gage is unaffected by whatever other loads may be imposed on the system.

The recording system is automatic upon command and contains features which are intended to reduce data processing to a minimum. The static-stability or steady-state data are obtained digitally and are recorded in tabular form by electric typewriter and also can be transcribed to IBM punch cards or recording tape. The oscillation measurements pass through a resolver and integrator and are recorded as essentially discrete values of in-phase and quadrature force components for each normalized frequency in pitching or heaving.

INTRODUCTION

The stability and control characteristics of a submerged body moving through a fluid can be understood best on the basis of a thorough analysis of the differential equations which govern the motion. These equations of motion are comprised of numerous coefficients or derivatives which are of hydrodynamic origin. Consequently, to obtain solutions for any given configuration it is necessary to know these coefficients with reasonable accuracy. Many attempts have been made in the past to fulfill this requirement by utilizing various experimental and theoretical techniques, or combinations of both.

Among the experimental methods used, fairly refined techniques have been developed by model basins and wind tunnels for measuring forces and moments due to hull orientation; the so-called static stability and control coefficients. However, the various experimental methods used to determine forces and moments associated with variations in angular velocity, linear acceleration, and angular accleration have been less successful. The techniques that have been tried in this respect include facilities such as the rotating arm, free oscillator, forced oscillator, curved-flow tunnel, and curved models in a straight flow facility. Some of these facilities may eventually provide the required accuracy. However, the desired stage of refinement has not been reached due to problems such as instrumentation and model support.

The theoretical means employed to obtain hydrodynamic coefficients also have been inadequate. With bare-body configurations, theory has been used with reasonable success to compute coefficients such as added mass and added moment of inertia which are amenable to treatment on the basis of potential flow considerations. However, coefficients which are primarily due to viscous flow, such as "static" and "rotary" forces and moments, are not obtained reliably with existing theory. With configurations which include appendages such as control surfaces, decks, fairwaters, and propellers, the calculations based on existing theory become even more suspect.

With full realization of the apparent shortcomings in both experimental and theoretical approaches to this subject, the David Taylor Model Basin initiated a study of the problem under its Fundamental Hydromechanics Research Program. As an outgrowth of this study, it was decided that the most direct approach would be to acquire a facility which would provide by experimental means all coefficients required in the equations of motion for six degrees of freedom of arbitrary submerged body-appendage configurations. Accordingly, techniques were devised and a design for equipment to perform this function was initiated in October 1956. In June 1957, construction was completed and shortly thereafter the new device called the DTMB Planar-Motion-Mechanism System was placed into regular service.

This paper outlines the considerations leading to the basic concepts, sets forth the principles of operation, and describes the apparatus and instrumentation of the DTMB Planar-Motion-Mechanism System. A few typical curves are given to illustrate the kinds of end results that are obtained with the system.

GENERAL CONSIDERATIONS

The derivations and composition of the equations of motion have formed the subject of numerous text books and papers. For the purpose of this paper, therefore, only the general nature of these equations are considered. This is done to give some insight into the problems which must be faced in the design of experimental facilities for the evaluation of the equations.

The hydrodynamic forces and moments which enter into the equations of motion as coefficients are usually classified into three categories: static, rotary, and acceleration. The static coefficients are due to components of linear velocity of the body relative to the fluid; the rotary coefficients are due to angular velocity; and the acceleration coefficients are due to either linear or angular acceleration. Within limited ranges, the coefficients are linear with respect to the appropriate variables and thus may be utilized as static, rotary, and acceleration derivatives in linearized equations of motion.

It may be concluded from the foregoing classification, that the experimental determination of the coefficients of the equations of motion requires facilities which will impart linear and angular velocities and accelerations to a given body with respect to a fluid. For example, the usual basin facilities have carriages designed to tow models in a straight line at constant speed. Such facilities can be equipped to orient models in either pitch or yaw to obtain the static coefficients. However, more specialized types of facilities, such as rotating arm or oscillator, are required to impart the angular velocities that are necessary to obtain rotary coefficients. The oscillator type of facility provides also linear and angular accelerations so that the acceleration coefficients may be determined experimentally.

The choice of a suitable facility for determining hydrodynamic coefficients involves many considerations pertaining to accuracy, expediency, and ease of data analysis. A detailed treatment of these problems is beyond the scope of this paper. However, of primary concern is the degree to which the experimental technique involves explicit relationships and avoids the need for solutions of matrices. Also techniques which involve extrapolations should be avoided. To illustrate, a carriage which tows a model at uniform velocity in straight-line pitched or yawed flight is a direct and explicit means of determining static coefficients. Similarly, a rotating arm which tows a model at uniform angular velocity and tangential to the circular path at each of several different radii is a means for determining rotary coefficients explicitly. On the other hand, the use of the rotating arm to obtain static coefficients should be considered as an indirect procedure since the data must be extrapolated to infinite radius. The usual oscillator techniques are even more indirect and, at best require solutions of simultaneous equations to obtain rotary and acceleration derivatives.

Each of the techniques mentioned can be used most advantageously for obtaining one category of hydrodynamic coefficients. The straight-line towing carriage supplies only the static coefficients. The rotating arm supplies rotary coefficients directly and static coefficients indirectly. The oscillator supplies all three categories of coefficients, but all indirectly.

The foregoing considerations suggest the desirability of having a single system to determine explicity all of the coefficients required in the equations of motion for six degrees of freedom. To accomplish this

objective, it is necessary to develop a facility which can move a body through water with "hydrodynamically pure" linear velocities, angular velocities, linear accelerations, and angular accelerations in all degrees of freedom. This concept forms the basis of the DTMB Planar-Motion-Mechanism System.

PRINCIPLES OF OPERATION

The DTMB Planar-Motion-Mechanism System as it physically exists is described in detail in the next section. It is desirable, however, to consider first the principles underlying the operation of the mechanism so that the design concept can be generally understood. The system was designed primarily for obtaining hydrodynamic characteristics of deeply submerged bodies in either the vertical or horizontal planes of motion. It can be used as well to obtain vertical-plane characteristics for bodies operating near or on the water surface. In the interest of simplicity, however, the mode of operation applying to submerged bodies in the vertical plane will be used to describe the principles of the system.

The kind of motion for static coefficients is commonly used by wind tunnel and model basin facilities and, therefore, does not need to be explained in detail. The diagram in Figure 1 schematically represents this type of motion. The components are given with respect to a body-axis system with the origin at the center of gravity, CG.

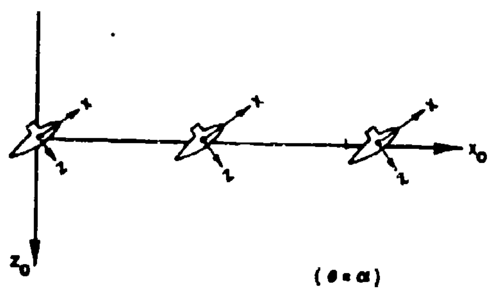
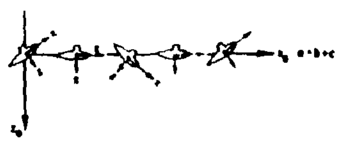


Figure 1 - Straight-Line Pitched Motion for Steady-State Tests

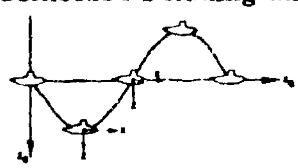
The system produces this motion by using a towing carriage to tow the model in a straight path at constant velocity. Discrete pitch angles for each run are set by a tilt table which supports the model through a pair of twin towing struts. Control surface angles are also set discretely for each run. Forces are measured by internal balances at each of the two struts to obtain static forces and moments.

The unique feature of the DTMB Planar Motion Mechanism is the kinds of motions produced to enable the explicit determination of the

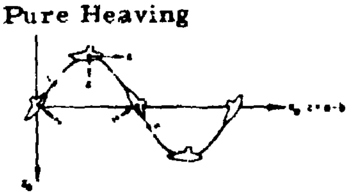
rotary and acceleration coefficients. Sinusoidal motions are imposed to the model at the point of attachment of each of the two towing struts while the model is being towed through the water by the carriage. The motions are phased in such a manner as to produce the desired conditions of hydrodynamically "pure heaving" and "pure pitching". It is possible also, if required for any reason, to produce various combinations of pitching and heaving. Figure 2 illustrates various types of motions including (a) the type of motion usually associated with oscillators, (b) pure heaving, and (c) pure pitching. The latter two are the basic motions associated with the DTMB Planar Motion Mechanism.



(a) Combined Pitching and Heaving



(b) Pure Heaving



(c) Pure Pitching

Figure 2 - Oscillation Types of Motion

The oscillator motion depicted by Figure 2a is actually a combination of pure pitching and heaving motions. The CG is constrained to move in a straight path while the model, which oscillates in a see-saw fashion, assumes sinusoidally varying angles of attack and pitch angles. Since the model is subjected to both linear and angular accelerations, a mixture of static, rotary, and acceleration forces and moments results. It becomes necessary, therefore, to perform a similar oscillation about a second reference point. The two oscillation conditions together with the static tests provide data which can be used to separate the hydrodynamic coefficients. The solution of simultaneous equations involved in this process, however, could lead to errors because of the wide differences in magnitude between the various individual coefficients. The oscillator type of motion is produced by the Planar Motion Mechanism when the two struts move sinusoidally at 180 degrees out of phase with each other.

The pure heaving motion shown in Figure 2b is obtained when both struts move sinusoidally in phase with each other. This results in a motion whereby the model CG moves in a sinusoidal path while the pitch angle θ remains zero.

The pure pitching motion shown in Figure 2c is obtained by moving both struts out of phase with each other; the phase angle between struts is dependent upon frequency of oscillation, forward speed, and distance of each strut from CG. The relationship is as follows:

$$\operatorname{Cos} \phi_s = \frac{1-\left(\frac{\omega x}{U}\right)^2}{1+\left(\frac{\omega x}{U}\right)^2}$$

where

ϕ_s is the phase angle between struts,

ω is the frequency of oscillation,

x is the distance of each strut from the CG, and

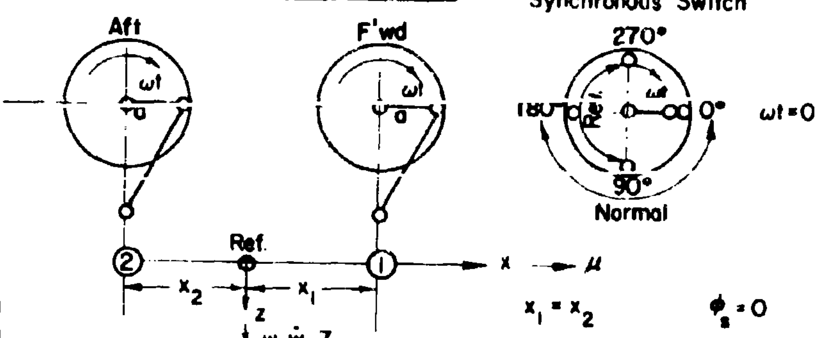
U is the forward speed of the model.

The resulting motion is one in which the model CG moves in a sinusoidal path with the model axis tangent to the path (angle of attack $a = 0$).

The process for obtaining translatory acceleration derivatives from pure heaving tests is represented diagrammatically in Figure 3. The diagrams across the top of the figure show the motions of the aft and forward struts with respect to each other. Corresponding positions of a synchronous switch, provided with the electrical system to rectify the sinusoidal signals from the force balances, are also shown. At the left is a column of graphs showing the resulting motions and forces at the CG. The right-hand column contains the mathematical relationships represented by each graph. Descending fr m the top of Figure 3, there is the vertical displacement z curve, the associated velocity $\dot{z}$ curve, the associated acceleration $\ddot{z}$ curve, and then the vertical force Z_R curve. It may be noted that the Z_R curve is displaced in point of time from the z curve by phase angle ϕ. Thus Z_R can be considered as being made up of two components, one in phase with the motion at the CG, Z_{in}, and the other in quadrature with the motion at the CG, Z_{out}. The shaded area per cycle under each curve represents the magnitudes of Z_{in}, and Z_{out}, respectively.

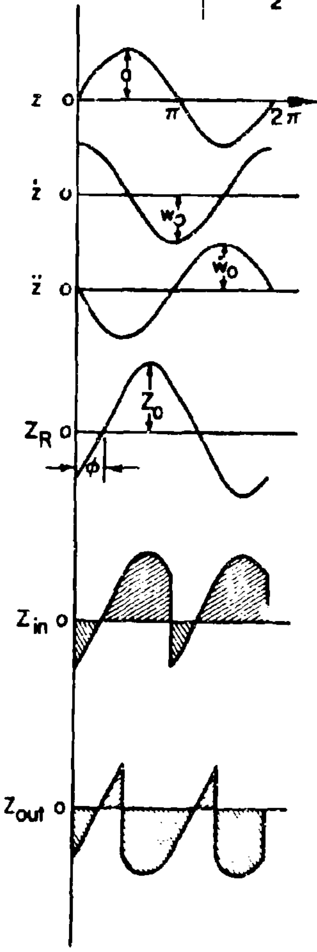
The in-phase component of force is directly related to the linear acceleration and, therefore, can be used to compute explicitly the associated acceleration derivatives. For example, the nondimensional acceleration derivative $Z_{\dot{w}}'$ which defines the added mass can be obtained as follows:

Pure Heaving



$x_1 = x_2 \qquad \phi_s = 0$

$x_1 + x_2 = b$



$$z = a \sin \omega t$$

$$\dot{z} = w = \omega a \cos \omega t = w_0 \cos \omega t$$

$$\ddot{z} = \dot{w} = -\omega^2 a \sin \omega t = -\dot{w}_0 \sin \omega t$$

$$Z_R = Z_0 \sin(\omega t - \phi) = (Z_0 \cos\phi)\sin\omega t - (Z_0 \sin\phi)\cos\omega t = Z_{in} \sin\omega t + Z_{out} \cos\omega t$$

$$Z_{in} = \frac{\pi}{2} \cdot \frac{1}{2\pi}\left[\int_0^{\pi} Z_0 \sin(\omega t - \phi)\, d(\omega t) - \int_{\pi}^{2\pi} Z_0 \sin(\omega t - \phi)\, d(\omega t)\right] = Z_0 \cos\phi$$

$$Z_{out} = \frac{\pi}{2} \cdot \frac{1}{2\pi}\left[\int_0^{\pi/2} Z_0 \sin(\omega t - \phi)\, d(\omega t) - \int_{\pi/2}^{3/2\,\pi} Z_0 \sin(\omega t + \phi)\, d(\omega t) + \int_{3/2\,\pi}^{2\pi} Z_0 \sin(\omega t - \phi)\, d\,\omega t\right] = -Z_0 \sin\phi$$

Figure 3 - Analysis of Pure Heaving Motion

$$Z_{\dot{w}}' = \frac{\partial[(Z_1')_{in} + (Z_2')_{in}]}{\partial \dot{w}_o} + m_m'$$

where

$(Z_1)_{in}$ and $(Z_2)_{in}$ are in the in-phase components at each strut of the resultant force Z_R,

$\dot{w}_0$ is the amplitude of the linear acceleration, and

m_m' is the mass of the model.

The process for obtaining rotary and angular acceleration derivatives from pure pitching tests is represented diagrammatically in Figure 4. The order followed is similar to that shown in Figure 3. In this case, the pitch angle traces (θ, $\dot{\theta}$, and $\ddot{\theta}$) are of primary interest. The Z_R curve is displaced in point of time from the θ curve by phase angle Φ. The procedure for resolving the resultant force into in-phase and quadrature components is similar to that for the pure heaving case. The shaded area per cycle under each curve represents the magnitudes of Z_{in} and Z_{out}, respectively.

In the pure pitching case, the in-phase component of force is directly related to the angular acceleration and the quadrature component is directly related to the angular velocity. Thus both the angular acceleration and rotary derivatives can be computed explicitly. For example, the nondimensional rotary derivative Z_q' can be obtained as follows:

$$Z_q' = \frac{\partial[(Z_1')_{out} + (Z_2')_{out}]}{\partial q_0'} - m_m'$$

where

$(Z_1)_{out}$ and $(Z_2)_{out}$ are the quadrature components at each strut of the resultant force Z_R and,

q_0 is the amplitude of the angular velocity.

The force components are measured separately at each of the two struts for both the heaving and pure pitching cases. Since the struts are equidistant from the model CG, all of the various moment derivatives associated with the oscillations are also obtained explicitly.

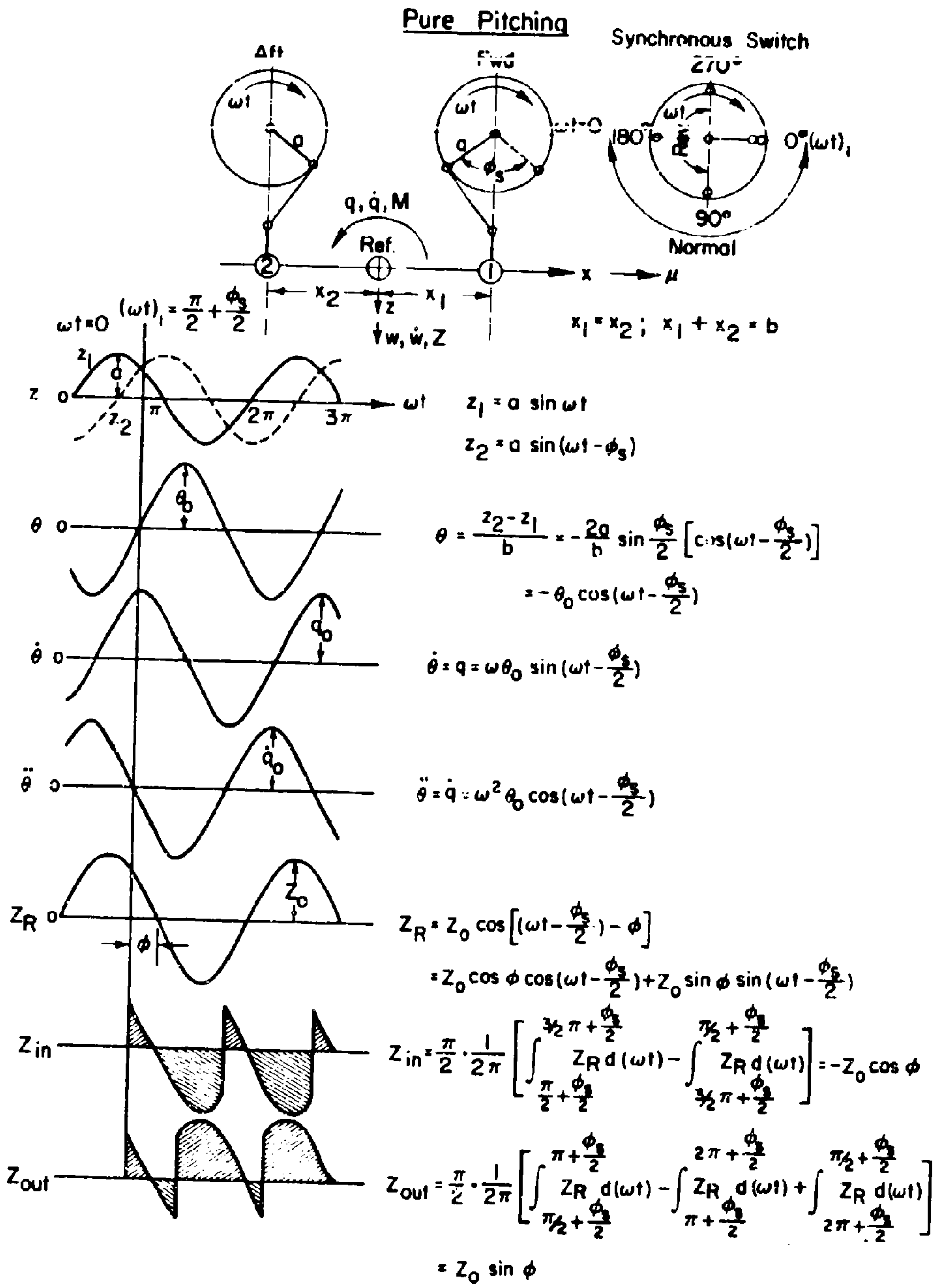


Figure 4 - Analysis of Pure Pitching Motion

DESCRIPTION OF APPARATUS

The DTMB Planar-Motion-Mechanism System is a complete system for obtaining hydrodynamic coefficients from model tests. It embraces all mechanical, electrical, and electronic components necessary to carry out all functions starting from the delivery of the model to finalized processing of data preparatory to analysis. This includes preparation of the model for testing, conduct of static and oscillation tests, sensing and recording of test data, and processing data digitally in tabulated form or for input to high-speed computers. The main features of the system are: model support and positioning equipment, forced-motion mechanism, dynamometry, and Instrumentation Penthouse containing recording and control equipment.

MODEL SUPPORT AND POSITIONING

Model support and positioning is accomplished by an assembly consisting of a tilt table and a pair of twin towing struts, as shown by the sketch in Figure 5. The assembly with model attached is portable, and can be moved about by an overhead hoist as shown in Figure 6. The portability is considered important at the Taylor Model Basin because of the heavy workload requiring the active use of the towing carriages.

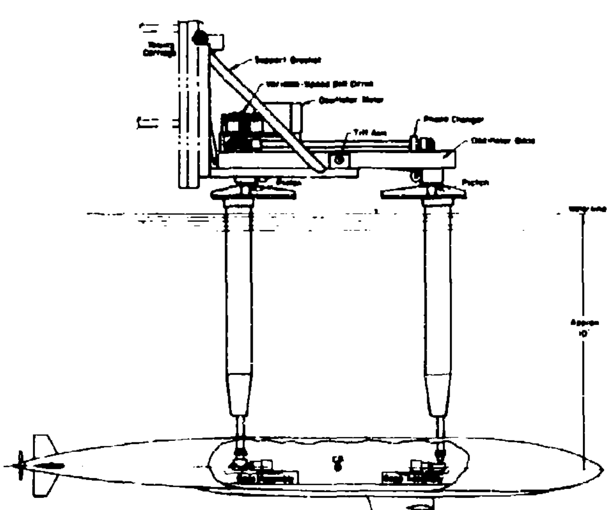


Figure 5 - Schematic Arrangement of DTMB Planar-Motion-Mechanism

When it is desired to rig, ballast, or make time-consuming changes, the assembly is placed on the storage stand shown in Figure 7. The storage stand is mounted on jacks and has reference surfaces for

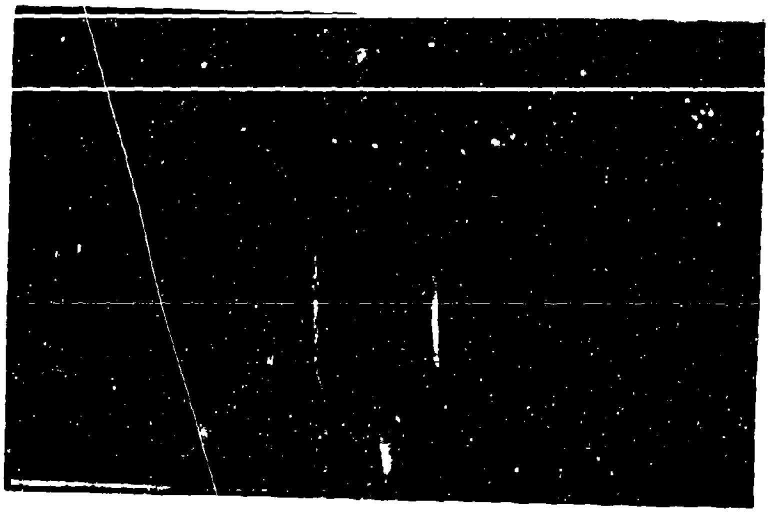


Figure 6 - Tilt Table with Model Attached Being Moved by Overhead Crane

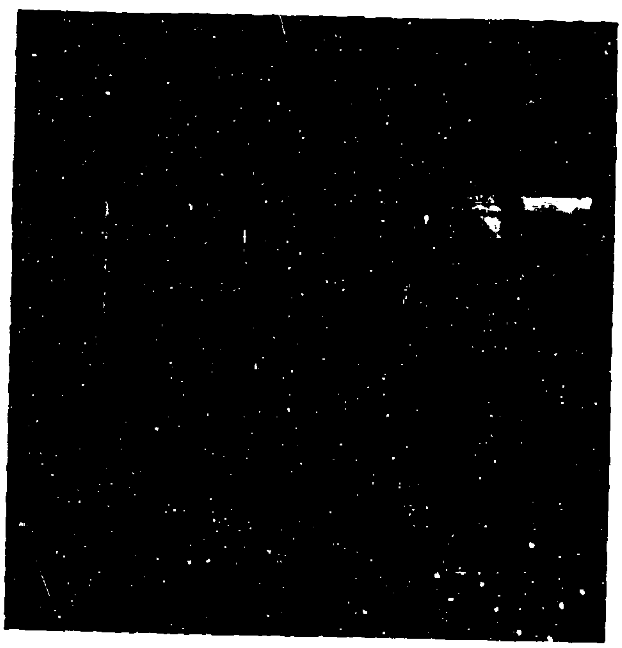


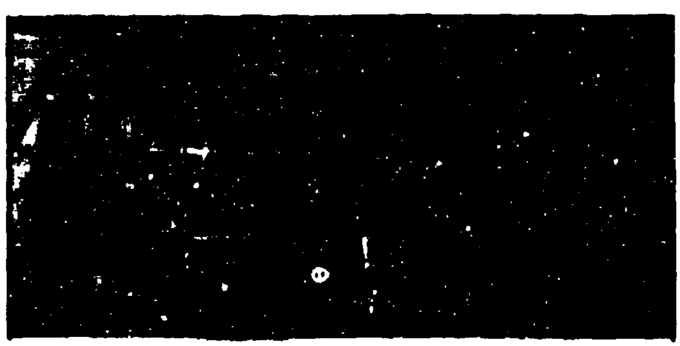
Figure 7 - Tilt Table with Model Attached Mounted on Storage Stand

levelling the model. Electrical power supplies and the Instrumentation Penthouse are accessible so that the model and instrumentation can be fully checked free and clear of the towing carriage. When used for testing, the assembly is mounted on a support bracket attached to and extending out from the towing carriage.

Tilt Table

The tilt table is shown schematically in Figure 5 and by the photographs in Figure 8. It is a rectangular frame constructed primarily of 8-inch steel I-beams welded together. The frame is about 9 feet 6 inches long and 2 feet 2 inches wide. A 3 7/8-inch diameter heavy-walled steel tubing is inserted transversely through the frame at the longitudinal midpoint and welded to it. The tubing serves as an axle for tilting the table in the pitch plane. The axle fits with close tolerance into a split-clamp trunnion bearing on the support bracket which is attached to the towing carriage. Caps are provided on the ends of the axle to prevent the tilt table from slipping sidewise.

(a) Elevation View



(b) Plan View

Figure 8 - Views of Tilt Table

Two 6 inch diameter vertical sleeve bearings (cylinders) are welded into the frame; one 45 inches forward and the other 45 inches aft of the tilt axis. The cylinders are steel and lined with bearing bronze. They are bored accurately to receive pistons which carry the towing struts. Each cylinder contains two keyways, placed 180 degrees apart, to maintain fore and aft alignment and prevent the pistons from rotating. A 3/4-inch hole is bored through each cylinder wall to receive

a steel centering pin which locks the pistons in place for the static stability tests and also serves as a reference point in the oscillation tests.

A machinery base about 10 inches high and 40 inches long is welded on top of the tilt table on the end facing the carriage. The base supports the oscillation motor and pulleys for the belt drive.

The table is tilted by the mechanism shown in close-up by Figure 9.

Figure 9 - Close-Up View of Tilting Mechanism

The end of the table is moved vertically by a 4-foot long Saginaw ball-bearing screw-jack mounted in the support bracket at the carriage end. A link which is pin-connected at each end joins the tilt table to the moving part of the screw-jack. A guide-rod installed in the support bracket parallel to the screw-jack provides additional stiffness against yawing or pitching of the tilt table. The guide-rod is a 2-inch diameter precision ground steel cylinder equipped with a sliding ball-bearing bushing. A fixture containing the bushing is connected to the moving part of the screw-jack. Thus the screw is always supported at the point at which the load is applied. Angular setting of the tilt table is accomplished automatically. The screw-jack is driven by a 1/3 hp, 550 volt, 3-phase electric motor. The gear reduction is such that the tilt table moves at the rate of about one degree per second. The motor is equipped with a brake to prevent coasting and is started and stopped by a command switch and a system of micro-switches installed on the support bracket. The micro-switches are each spaced approximately one-inch apart

amounting to exactly one degree on the tilt table. In this manner, discrete angles can be commanded from a remote station in one-degree steps over a range of ± 20 degrees.

The tilt table assembly also can be moved vertically to change the depth of submergence of the model. This is accomplished by raising or lowering the support bracket which rides up and down on a pair of parallel vertical rails attached to the carriage. An electric hoist mounted on top of the towing carriage provides power for the movement. The hoist cable is hooked to a pad-eye on the top of the support bracket. When a vertical movement is not desired, the support bracket is clamped tightly to the rails with quick-release clamps.

Towing Struts

A twin strut system was adopted as the method of towing submerged models. This decision was reached on the basis of thorough studies of the towing problem including hydrodynamic, structural, and handling aspects. In the design of strut systems for towing bodies that are apt to be unstable, the torsional rigidity of the system must be made to exceed the anticipated static-moment rate of the model in yaw, pitch, and roll. The torsional rigidity of a twin strut system in pitch and yaw can be made greater by increasing the spacing between the points of attachment of the struts. Thus for equal torsional rigidity, a much larger section is required for a single strut than for one of the twin struts. It is of utmost importance to make the size of the strut small in proximity to the model to minimize strut interference effects. Consequently, the twin strut system is at a decided advantage in this respect. Also, it is more feasible to make a twin-strut system stiff enough so that the angles set at the unloaded condition will remain essentially the same while the model is being towed at maximum speed and high angle of attack.

The strut arrangement for the Planar Motion Mechanism can be seen in Figures 5 and 6. The struts are attached to the tilt table through the pistons. A clamp between the piston and top of the strut allows for attachment and adjustment of the spacing between the struts. The adjustment is made by a hand-driven worm screw which moves the strut relative to the piston. The strut spacing can be varied from 4.5 to 9.0 feet.

The present strut is of simplified construction and was designed from the standpoint of economy and ease of fabrication. Nevertheless, the portion of the strut in proximity to the model was carefully designed from a hydrodynamic standpoint. In the future these struts will be replaced by a set of more sophisticated design. The struts basically consist of an upper part, a transition, and a lower or small part. The upper part is about 7 feet long and consists of an internal strut or core to which an external fairing is attached with machine screws. The core has a 7- x 1 3/4-inch rectangular section and is constructed of 1/4-inch stainless steel plates which are bent into angles and welded at adjacent corners. Intermediate stiffeners running lengthwise are welded inside

of the rectangular section. The fairing is 1/8-inch sheet aluminum which is bent in two halves to form a simplified hydrofoil-shaped section. The middle part of the fairing section is parallel and conforms to the core. The leading and trailing edges of the fairing are welded together. The outside of the upper part of the strut thus has a uniform section about 12 inches on chord and 2 inches thick. A gusset plate welded to the top of the strut serves as the means of attachment to the pistons on the tilt table.

The transition part of the towing strut is constructed similarly to the upper part but the core and fairing are tapered so that in a length of 20 inches the outside section tapers down to a chord of 6 1/8 inches. Both the core and fairing of the transition are welded to the upper part of the strut. The bottom of the transition contains a split clamp which holds and permits adjustment of the lower part of the strut.

The lower part of the strut was deliberately made as small as practicable to minimize strut interference effects. Externally, it has an ogival section 3 inches on chord and 1 1/8 inches thick. It is 36 inches long and can either be retracted within the upper part of the strut or adjusted up to an extension of 30 inches. The small part of the strut was constructed by rolling two 1/8-inch stainless steel plates into circular arcs and welding them together at the leading and trailing edges. It was then heat treated and precision machined to obtain accurate alignment of the model and strut when held in the clamp. When testing, part of the small strut is within the model. Therefore, for the bottom 1 3/4 inches, the trailing edge of the ogival section was opened up in a U-shaped fashion to facilitate passage of electrical cables through the strut. A disk-shaped pad welded to the bottom of the strut permits attachment by bolts to the dynamometers within the model.

In addition to the use of small strut sections in proximity to the model, the method of setting hull angles also strongly minimizes strut interference effects. When an angle is set on the model, the struts rotate in the vertical center plane and thus maintain a zero angle of attack with respect to the flow. The interference effect is largely due to lift induced on the hull by the struts and since the struts remain at zero angle of attack, this type of interference effect is not present. It has been found that the interference effects on lift and moment with this strut system are small enough to be neglected for models as small as 9 feet in length. The effects on drag which are due primarily to the wake left by the struts are also very small.

FORCED-MOTION MECHANISM

The motions for the oscillation tests are supplied by a forced-motion mechanism mounted on the tilt table. The mechanism consists of an electric motor which utilizes positive drive pulleys to drive a slider-crank attached to each of the two strut-pistons. The slider-cranks are joined by a common drive-shaft which passes through a phase-changing device. A counterbalancing device is provided for overcoming the

deadweight load of the moving parts.

Drive System

The drive system is shown by the close up view in Figure 10. The prime mover is a 1 hp, 3-phase, 60-cycle, 550 volt General Electric motor. The motor contains a planetary gear that reduces the speed by a ratio of 85 to 1. Thus the speed at the output shaft is 20 rpm.

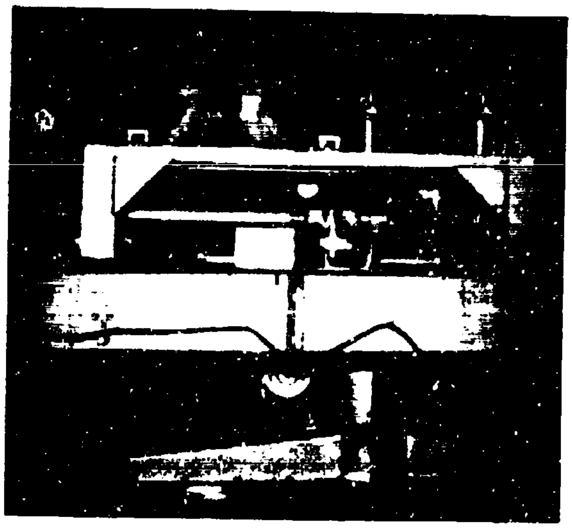


Figure 10 - Close-Up View of Drive System for Forced-Motion Mechanism

The fixed-speed motor was chosen because it provides maximum efficiency for minimum weight since the motor always runs at its rated speed. Another consideration was the desire to avoid the use of variable-speed-control systems. Such systems, whether they be frequency control for alternating current motors or amplidyne control for direct current motors, are very expensive and can be troublesome. The characteristics of the induction type motor are reliable and the motor will maintain constant speed with precision when subjected to the variable loads expected from the tests. This factor has been thoroughly verified by speed calibrations with the mechanism subjected to such variable loads.

The variable-speed feature is not needed for submerged bodies since, as mentioned earlier, the normalized frequency varies directly with oscillation frequency and inversely with towing carriage speed. The carriage has precision control over a continuous range of speeds and can, therefore, be used as the basis for the normalized frequency change.

There are cases where it is desirable to have the drive mechanism supply more than one discrete oscillation frequency. For example, such a feature is helpful in standstill runs or where it is not feasible to cover

a broad enough range by varying carriage speeds. It is also helpful when the technique is used to obtain the moment of inertia of the model in air. Thus, the pulley system shown in Figure 10 was incorporated into the design to provide a selection among three discrete frequencies of rotation. The system utilizes positive-drive pulleys and a Gilmore timing belt to cause the drive shaft to rotate at 1/2, 1, and 1 1/2 times the speed of the motor output shaft. This results in oscillation frequencies of about 1.1, 2.2, and 3.3 radians per second. To avoid speed variation, the pulleys used were of a special anti-backlash type. The belt is changed from one pair of pulleys to another by lowering or raising the motor shaft with respect to the drive-shaft. This is accomplished by tilting the motor platform about a hinge along its edge. The tilting is done automatically with two small motorized screw-jacks. This device enables the changes to be made quickly and provides a good means for adjusting the tension in the belt.

The drive-shaft is 1 7/8-inch-diameter steel tubing which runs the length of the tilt table. It is supported near each end by pillow blocks. Thrust bearings are provided to restrain the drive-shaft longitudinally.

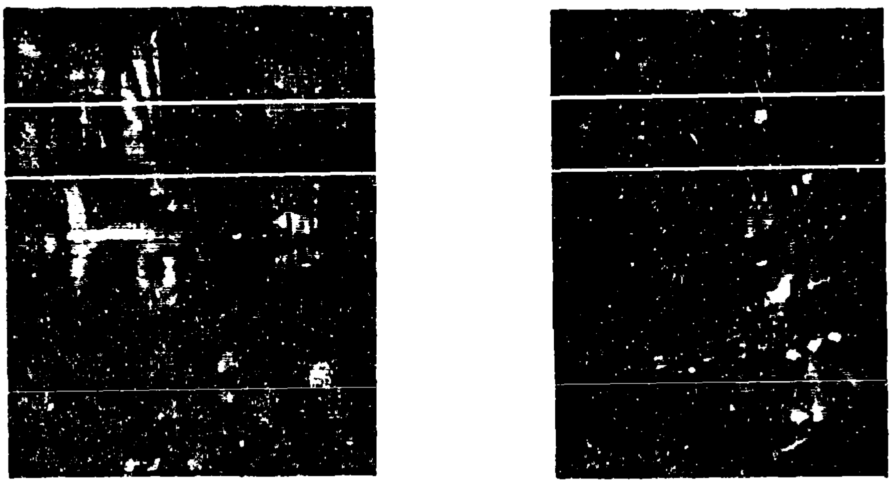
Slider-Crank System

In the interests of simplicity and reliability, a slider-crank mechanism, instead of a skotch yoke, was used to obtain sinusoidal motion at each strut. The crank-arm which provides a 1 inch eccentricity is mounted on the drive-shaft above the centerline of the piston. The connecting rod is attached to the crank-arm and a wrist pin in the piston. It is 17 1/8 inches long between pin centers. The piston is constrained, with close tolerance, to linear motion within the cylinder. It is keyed to prevent rotation about its longitudinal axis.

Since the ratio of length of connecting rod to eccentricity is 17.125, the resulting motion is within 1.5 degree of being truly sinusoidal. There is no appreciable error introduced by assuming that the motion is sinusoidal for purposes of analysis.

Phase-Changer and Synchronous Switch

The type of motion imparted to the model, whether it is pure heaving, pure pitching, or some combination of the two, depends upon the phase relationship between the motion of the two struts. The phase angle is established by the phase-changer shown by the close-up view in Figure 11. The phase-changer is essentially an index head consisting of two aluminum disks which serve as flanges to bolt together the drive-shaft at a convenient point between the slider-cranks. One disk contains calibrations every 1 degree about the circumference for 360 degrees; the other a vernier index for resolving the angle within 0.1 degree. Between the disks is a worm-drive operated by a removable hand crank which rotates one disk relative to the other. A phase change is made



(a) Top View (b) End View

Figure 11 - Close-Up Views of Phase-Changer and Synchronous Switch

loosening the three bolts holding the disks together, turning the worm-drive to obtain the prescribed angular setting, and then tightening the bolts. When the phase-changer is set at zero, the centering pins can be inserted to lock the struts in place for static tests, or if the pins are removed, the mechanism will provide pure heaving motion. If pure pitching motion is desired, the phase-changer is set at a predetermined angle which depends upon the oscillation frequency, strut spacing, and carriage speed, as mentioned earlier.

The end results sought in the oscillation tests are the separate force and moment components which are either in phase or in quadrature with the input motions. To accomplish this objective directly, an electrical system which resolves the sinusoidal signals coming from the force balances into in-phase and quadrature components is made part of the test equipment. The "brain" of the resolving system is the synchronous switch shown in Figure 11 which simultaneously selects either the in-phase or quadrature parts of the signals coming from all of the force balances.

The synchronous switch assembly consists of a bearing support mounted on the tilt table platform and a rotating drum connected by an Oldham coupling to the drive-shaft. Mounted on the face of the bearing support are four micro-switches. The rollers which actuate the switches are spaced 1.875 inches from the center of the shaft. The micro-switches are set accurately so that they are tripped at exactly 0, 90, 180, and 270 degrees. The micro-switches are tripped by a ball-bearing-tipped sweeper mounted normally to the drum periphery. The sweeper length is screw-adjusted and locked into place to carefully control the pressure on the switches. The rotating drum is calibrated in one-degree

increments about 360 degrees. A vernier index mounted on the bearing support permits setting of angles to within 0.1 degree. The drum can be rotated with respect to the drive-shaft by releasing a split-clamp which holds it to its own shaft.

As with the phase-changer, the setting of the synchronous switch must be altered to conform to the kind of motion being produced. For pure heaving, the procedure is straight-forward. The pistons are set in mid-position corresponding to a setting of zero on the phase-changer. The centering pins are inserted through the cylinders to hold alignment. Then, by releasing, rotating, and tightening the clamp, the actuator of the synchronous switch is set to zero position as indicated on the drum scale. For each condition of pure pitching, it is necessary to reset the switch actuator to a new position. There are various techniques for doing this, but each amounts to indexing the drum on the synchronous switch to one-half the angle set on the phase-changer.

Counterbalancing

In addition to overcoming hydrodynamic loads, the drive-motor of the forced-motion mechanism must raise and lower the unsupported deadweight load of the moving parts of the system. Assuming a neutrally buoyant model, this load is caused by the weight of connecting rods, pistons, strut supports, struts, and part of the gage assemblies. The deadweight would normally impose a sinusoidal load on the drive-motor of considerably greater amplitude than the maximum hydrodynamic load anticipated. Consequently, the use of alternative systems of counter-balancing was investigated. Counterbalancing weights were discarded for two reasons, first, the weights would substantially increase the total weight of the system to be cantilevered on the support bracket and secondly, there would be problems of restraining the weights from swinging to avoid inertial effects. The system shown by the close-up in Figure 12 was adopted, therefore, as the means of counterbalancing.

Figure 12 - Close-Up View of Counterbalancing Device

It may be seen that the system is made up at each strut of eight individual flexator springs joined together and attached at the strut by a pulley arrangement. A flexator is a type of spring which exerts nearly constant tension over its design range of deflections. Its net effect, therefore, is very similar to the use of weights. Each of the flexator springs used has a capacity of 50 pounds. Thus, the total system counterbalances 800 pounds but only weighs 50 pounds itself.

DYNAMOMETRY

The dynamometry is composed of a system of gages designed to measure forces and moments in six degrees of freedom. The gages are installed within the test model as shown in Figure 5. An internal gage system was chosen in preference to the external types which are commonly used in similar wind-tunnel applications for the following reasons:

1. It eliminates the need for strut-tare corrections or, in the alternative, housing the towing struts within fairing. The latter technique is undesirable since it tends to increase the overall section size of the strut in proximity to the model and thus aggravates the problem of minimizing strut interference effects.

2. The balances are fixed to and rotate with the model so that the forces and moments are always measured with respect to the body axes. This is considered to be the preferred end result for analysis of the coefficients in the equations of motion.

The major components of the system are the modular force gages and the roll gages. The individual components and how they are assembled within the model to operate as a system are discussed in order.

Modular Force Gage

A modular type of gage was adopted as the basis for providing a force and moment measurement system which is free of interactions both mechanical and electrical. It is well known that other types of flexural multi-component balances suffer from mechanical interactions, that is, indirect loads affect the strains or deflections that are being measured. Attempts are made, with varying degrees of success, to mask this effect by arrangement of electrical transducers. A typical technique is the use of rosettes with bonded wire-resistance strain gages. Interactions are particularly objectionable for two reasons:

1. They affect the accuracy of a system especially where the combined loads are large compared with the direct load being measured.

2. They require the use of matrix-type calibrations. This requirement adds greatly to test preparation and data reduction time and is also cumbersome during testing where essentially end results are desired for plotting and checking purposes.

The modular force gage used with the system is shown in Figure 13. It is cube-shaped, 4.000 inches on edge and machined out of a solid block of ARMCO Steel Corporation 17-4 PH stainless steel. This material was selected for its excellent flexural properties, its corrosion resistance, and because it can be finished-machined and heat-treated without distortion or warpage. It has practically zero mechanical hysteresis; within the accuracy of measurement, the load-deflection curve is the same in both loading and unloading.

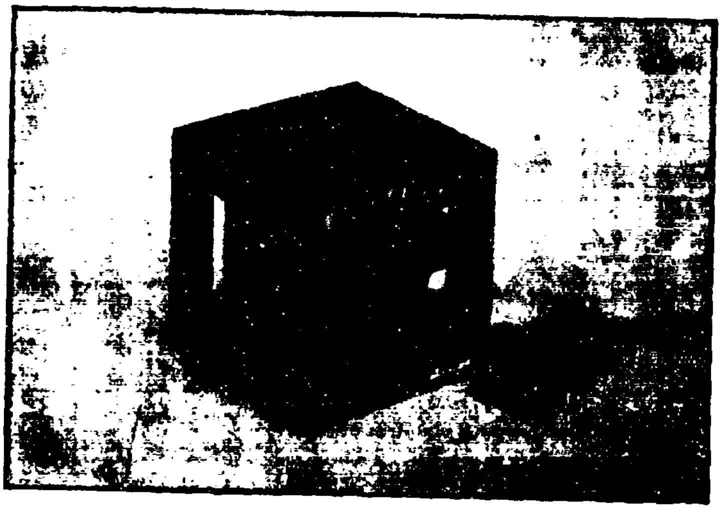


Figure 13 - Modular Force Gage

It may be seen that the cube has three pairs of faces of different type, designated as flexures, mounting surfaces, and open ends, respectively. Each flexural face is composed of two flexures 2.500 inches long, 0.186 inches thick, and 1.00 inches wide. A 2.0-X 2.5-inch rectangular opening allows access to the transducer within the gage. Each mounting surface has 4 holes, one at each corner, which are tapped to receive 3/8-inch bolts and 2 holes arranged near opposite edges on one centerline which are drilled and reamed to receive 1/4-inch aligning pins. Both mounting surfaces are identical except that one contains an additional hole to be used with the stop. The open ends of the gages are made up of the thickness of the flexures and mounting surfaces. The dimensions of all gage units are made identical to provide interchangeability.

The inside of the gage is formed by machining away as little material from the cube as practical. This was done to retain simplicity and to allow for very rigid support members. The two major parts within the gage block are the pedestals which support and maintain relative position of the transducer coil and core. The coil pedestal is an unusually stiff member which is an integral part and moves with one mounting surface. It holds a 1 1/8-inch diameter cylindrical stop which projects through the opposite mounting surface and flush to the outer face. The stop, limits the amount of gage travel and thus guards against overload of the flexures. It has a clearance of 0.031 inch on diameter.

The core pedestal is also rigid and is part of the mounting surface which moves opposite to that for the coil. A tapped hole in the pedestal provides screw-adjustment of the core relative to the coil. A lock nut holds the core piece in place after adjustment is made. With the foregoing arrangement, the transducer senses the deflection of the flexures as a parallel movement of one mounting surface relative to the other. The movement is equated to load on the basis of a static calibration with weights.

The spring constant of the flexure boxes was chosen high enough to obtain a natural frequency which would not result in magnification of oscillatory forces due to either carriage vibrations or forced-mechanism motions yet low enough to obtain good sensitivity and resolution. The relative deflection of the gage mounting surfaces is about 0.010 inch for a load of 500 pounds. The natural frequency with a 2000-pound model attached is about 30 cycles per second for each gage compared with a maximum frequency of 0.5 cycle per second for the forced motion mechanism. It is apparent, therefore, that the static calibration applies with rigor to the oscillatory forces measured.

The flexure boxes are exceedingly stiff with respect to forces and couples normal to the mounting surfaces and open ends. The only possible source of mechanical interaction, therefore, would be a large force exerted at the center of the mounting surfaces while the flexures are inclined. The movement of the flexures has been kept down to 0.01 inch for a direct load of 500 pounds. Consequently, an indirect load of 500 pounds would cause an interaction of about 0.05 percent, which is less than can be detected with most existing calibration devices.

The transducer used with the force gage is the so-called TMB Magnigage. The magnigage used is wound as a variable-reluctance type gage. It is similar in some respects to commercially obtainable units, however, it is magnetically shielded and is potted in plastic to operate efficiently when completely immersed in water. A watertight disconnect especially developed for the purpose is used for electrical connection to the recorders.

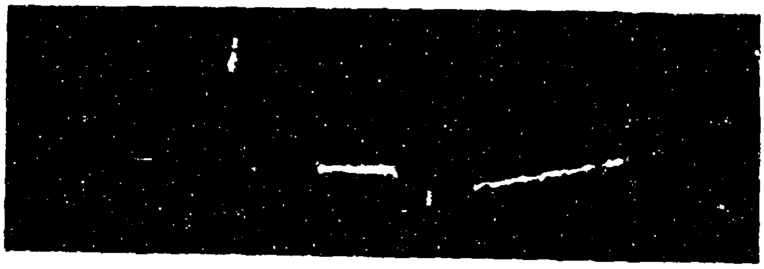
The electrical signal coming from the transducer changes when the core is displaced axially relative to the coil because of changes in length of air gaps between poles. The sensitivity as well as the range of

linearity of the transducer is governed by the ratio of maximum core movement to length of air gap. This ratio is usually predetermined on basis of maximum deflection and attendant maximum load anticipated for the gages. The cores presently installed within the transducers were selected to give optimum characteristics over a range of ± 0.004 inch. Taking into account the spring constant and the means for adjusting electrical sensitivity discussed later, each modular force gage is calibrated and adjusted to give a reading on the digital recorder of exactly 1000 counts for 200 pounds force. Since the calibration is linear, each count is equivalent to 0.2 pound. The electrical sensitivity can be changed to extend the range of measurement if desired.

Roll Gage

The modular force gages provide the means for measuring all required forces and moments except roll moment. Separate gages to measure roll moment are needed, therefore, to complete the system. The transverse sections at point of strut attachment on most models were not large enough to accommodate an offset modular force gage. Consequently, a different type of gage was selected for this purpose. The roll gage shown in Figure 14, is designed to measure pure torque about one axis in contrast to the modular gages which measure one component of pure force. It is not affected by the forces and moments exerted in other directions.

(a) Assembled with Gimbal



(b) Individual Components

Figure 14 - Roll Gage

As seen in Figure 14, the roll gage is very similar to the TMB Magnitorque units used with the transmission propulsion dynamometers. The primary elements of the gage are the shaft assembly and coil unit. The two main features of the shaft assembly are the flexural section and the armature. The shaft itself is K-monel metal. The flexural element, located about midway along the length of the shaft, consists of two sections, each about 1.125 inches long, which are necked-down from a diameter of 1.370 to 1.200 inches. The two sections are spaced 3/8 inch apart and are spool-like in appearance. The K-monel was selected as the flexural material because of its low mechanical hysteresis and non-magnetic properties. The armature is magnetic stainless steel and consists of three rings which are fastened over the shaft, one at each end and one at the middle of the flexural unit. The configuration of the rings is such that four longitudinal air gaps are formed at each side of the center ring. When a torque in a given direction is applied to the shaft, the air gaps on one side decrease and those in the other side increase by an equal amount. The differential changes in magnetic path cause the signal changes in the transducer.

The shaft was turned down from larger stock to provide a 3 9/16-inch diameter flange at one end for mounting purposes. The flange is bolted to the model through the chain of force gages. The other end of the shaft is attached to a gimbal block, and in turn the strut, by two number 8 taper pins rotated 90 degrees with respect to each other.

To maintain reasonable stiffness in roll, the torsional spring constant was selected as 650 pound-feet to produce a twist angle of 30 minutes. The air gaps in the armature were made 0.01 inch to obtain the desired sensitivity and linearity characteristics.

The coil unit used in conjunction with the armature constitutes the transducer for the roll gage. The coil used is wound as a variable reluctance type gage. The coil is potted in plastic so that it will operate efficiently when fully immersed in water.

The roll gage is inserted and fastened into a block. The block is also part of a gimbal which allows freedom about the pitch and yaw axes but offers restraint about the roll axis. Thus when a roll moment is applied, there is essentially no rotation of the coil or the fixed end of the shaft.

Taking into account the spring constant and the means for adjusting electrical sensitivity, each roll gage is presently calibrated and adjusted to give a reading on the digital recorder of exactly 1000 counts for 200 pound-feet of moment. Since the calibration is linear, each count is equivalent to 0.2 pound-foot. The electrical circuitry for the roll gage is similar to that of the force gages; the electrical sensitivity can be changed to extend the range of measurement if desired.

Gage Assembly and Arrangement

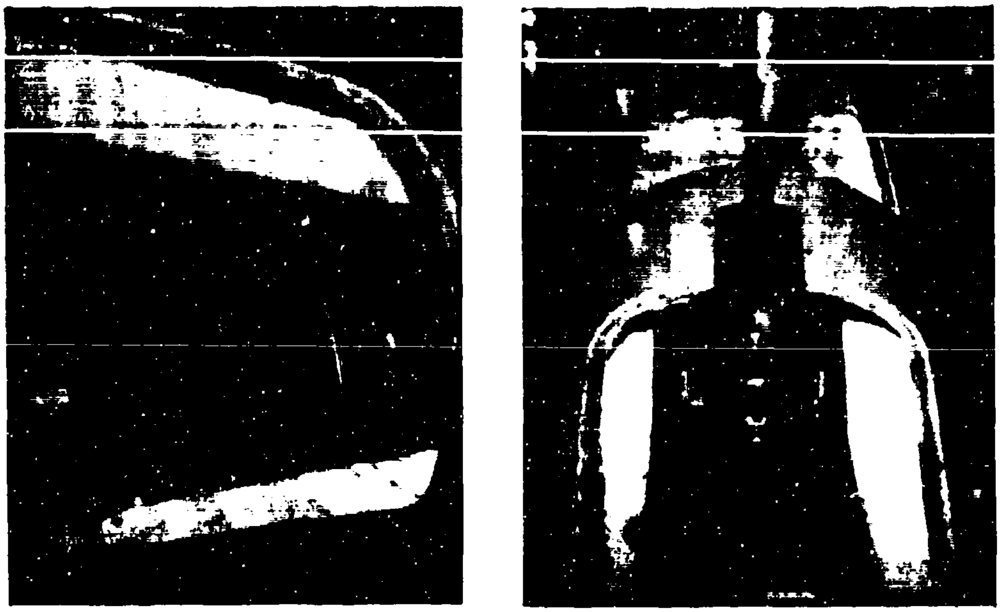
It is possible to vary the number, orientation, and arrangement of the individual gages to tailor the measurement system to the requirement of a specific test. The standard arrangement used for testing submerged bodies with the DTMB Planar-Motion-Mechanism System is shown in Figure 5. It is seen that the gage assembly at each of the two struts is connected from model to strut by a gimbal which allows freedom about its own pitch and yaw axes but offers restraint about its roll axis. The centers of the two gimbals are aligned with the body x-axis and are equidistant from the origin which is usually taken as the prototype CG scaled down to model dimensions.

It is convenient to resort to the analogy of a simply supported beam to explain how forces and moments are determined with the foregoing arrangement. Taking for example motions in the vertical plane, the total Z-force exerted on the model (beam) is experienced as pure reaction forces at each of the gimbal centers; the moment about each of these centers is zero. The reaction Z-forces are measured by the gages and their vector sum is equal to the total Z-force. These reaction forces are then resolved with respect to the CG to obtain pitching moment. Because of symmetry, the pitching moment is the vector difference between the reaction Z-forces times the distance from one gimbal center to the CG. The total X-force exerted on the model is also experienced as reaction X-forces which are measured by the gages at each of the two gimbal centers. The vector sum of the reaction X-forces is equal to the total X-force but since the reaction X-forces are aligned with the axis, there is no contribution to pitching moment.

The same technique is used to obtain the forces and moments in the horizontal plane leaving only the roll moment to be determined. The total roll moment is equal to the sum of the reaction roll moments which are measured by the roll gage at each strut.

It may be noted that it is possible to eliminate one roll gage and one X-gage and yet measure all required forces and moments. To do so, a gimbal with three degrees of freedom and a slider which permits movement along the x-axis are installed at one strut. This allows the entire roll moment and X-force to be taken on one roll gage and one X-gage, respectively. It was considered desirable, however, to have a closed elastic system and thus avoid problems of friction and lost motion which have harassed some of the earlier two-strut systems.

The gage assembly at each strut can be examined more closely in Figure 15. The model is equipped with a bedplate to receive the gage assembly. The first force gage is bolted to the bed plate through a mounting plate; its flexures face the x-direction so that it measures the X-force. The opposite mounting surface of the first gage is bolted by a channeled plate to one mounting surface of a second gage whose flexures face the y-direction so that, it measures Y-force. The other mounting surface of the Y-force gage is bolted by a gusseted angle bracket to a



(a) Top View (b) Fore and Aft View

Figure 15 - Gage Assembly

third gage whose flexures face the z-direction so that it measures Z-forces. The remaining mounting surface of the Z-force gage is bolted to the shaft of the roll balance which is connected through the gimbal to the towing strut pad. Thus, starting at the model, the order followed in the chain of gages is: X-force, Y-force, Z-force, and roll moment.

INSTRUMENTATION PENTHOUSE

A portable room called the Instrumentation Penthouse contains all of the electrical recording, control, and readout equipment for the DTMB Planar-Motion-Mechanism System. The Penthouse is a steel framework made up of I-beams and channels. It has a 10-X 14-foot decked-over floor; the four walls are 7-feet high and covered with lucite except for an open doorway and the ceiling is open.

The Penthouse can be moved by overhead crane either onto the carriage for test purposes or to a convenient storage space for calibration and checking of instruments and model. When testing, the Penthouse, as the name implies, is mounted on the top of the carriage and overlooks the model and other equipment, as shown in Figure 16. It serves as the center of operation for the engineers involved in the test. In addition to test instrumentation, the Penthouse contains desks, other office equipment, and fluorescent lighting so that calculations, working

plots, and even finalized analyses can be made on site.

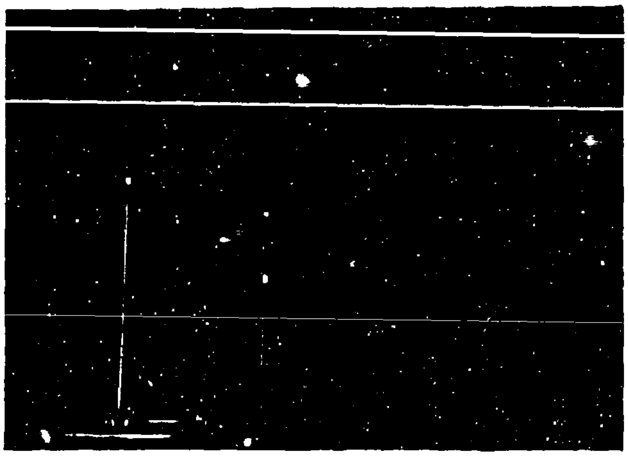


Figure 16 - Penthouse Mounted on Top of Carriage

All electrical circuitry which comes from the model and forced-motion mechanism to the Penthouse is joined together by a gang-disconnect. The female half of the disconnect is mounted on the tilt table and the male half is attached to the cables which remain with the Penthouse. This device is an effective means of saving hook-up t me and also serves to minimize errors in wiring which so often arise under the stress of meeting a test schedule.

The instrumentation within the Penthouse is shown in Figure 17.

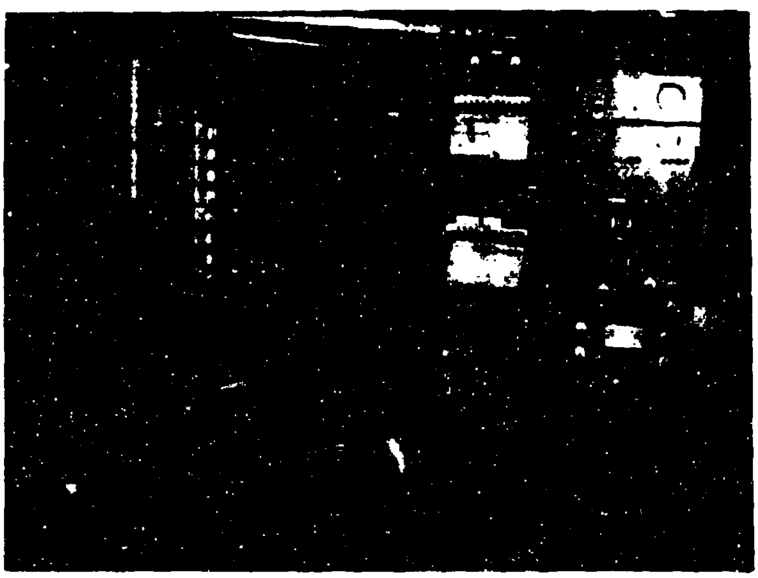


Figure 17 - Inside View of Penthouse Showing Instrumentation

It consists of three general categories: recording system for steady-state tests (statics), recording system for oscillation tests, and programming and control equipment.

Recording System for Steady-State Tests

The recording systems for both the steady-state and oscillation tests are alike in many respects and actually use common components. In interests of clarity, however, the two are separately described as complete systems.

The steady-state recording equipment is a digital system designed to display and readout the unique steady-state value of each force and moment sensed by the transducers for any given test condition. The equipment is contained within the two racks on the left-hand side of Figure 17. The typewriter shown in the figure is also part of the digital system. It may be seen that the system is made up of 8 channels to conform to the number of gages in the model. Each channel is separate in all respects except for the power supply that it shares in common.

Briefly, each channel is essentially an automatic null-balancing system; the transducer in the gage and the digital indicator combine together in a servo system. The transducer output is balanced by a potentiometer. When a gage in the model is deflected, the resulting error signal from the transducer is amplified and drives a servo motor which positions a potentiometer to restore electrical balance, or null, to the system. The amount that the potentiometer is moved is then a measure of the force or moment applied at the gage. The various components and circuitry which constitute the recording systems are shown by the block diagram in Figure 18.

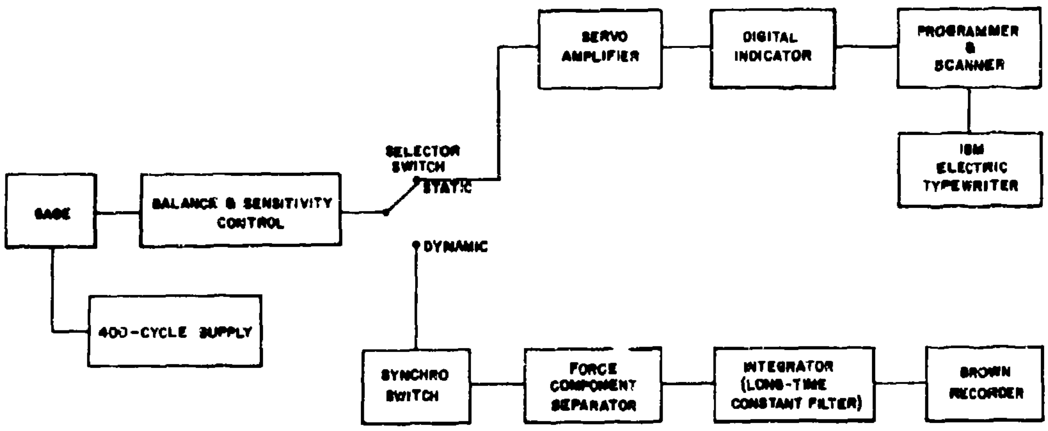


Figure 18 - Block Diagram of Recording Systems

The upper route of Figure 18 applies to the digital system used for steady-state tests. The term gage is used to denote the variable reluctance transducer whether it be the magnigage used with the modular force gages or the magnitorque used with the roll gages. The 400-cycle power source supplies a 4.5-volt carrier to the gage in such a manner that the current divides into two paths, one about each coil. If the core of the gage is electrically centered, the impedances of the gage halves are equal and, consequently, the voltages are equal. As the core is displaced, the impedance of one gage half increases and that of the other decreases with corresponding voltage changes.

Alternating voltages from the gages pass to the balance and sensitivity control box which contains two silicon diode bridges as well as other adjustments and refinements that are described in more detail later. These voltages are rectified by the diode bridges to produce full-wave rectified direct current voltages. The total rectified voltage obtained across both coils is constant and is used as a reference voltage. Polarity is established by making one side of the line positive and the other negative. The voltage measured between each coil changes, however, when the gage core is displaced; the voltage across one coil increases while the other decreases to an equal extent so that the reference voltage always remains constant. This is analogous to a three wire system in which the voltage across the outside lines remains constant but the voltage from one side to the common is made variable. The feedback potentiometer, which is on the shaft of the digital indicator, is wired similarly; the voltage across the end terminals is the reference voltage and the common is attached to the potentiometer slider. When the gage core is at electrical center, the potentiometer slider is at mid-position. When the gage core is displaced, an error signal results.

The error signal is fed to a chopper servo-amplifier similar to that contained in the Brown Recorder manufactured by Minneapolis-Honeywell Company. The chopper converts the direct-current error signal into 60-cycle alternating current. The resulting signal is amplified to drive a servo motor which in turn drives the potentiometer slider until the error signal is reduced to zero and a null-balance established.

The digital indicator, shown in Figure 19, is the active part of the feedback loop. The assembly is made up largely of commercially obtainable components. Beginning from the left, it may be seen that there is a digitizer, detent unit, servo motor, speed reducer, and potentiometer. The components are aligned axially and are connected together with Oldham couplings to minimize binding. The Metron speed reducer, is an anti-backlash planetary gear box with a reduction of 21 to 1. It is inserted between the servo motor and 10-turn Heliopot (± 0.05 linearity) so that the range of the system is ± 5 turns on the potentiometer, with a little to spare. The digitizer is connected directly to the through-shaft of the servo motor. The digitizer, called a Digicon, is a unit manufactured by Anatran Engineering Corporation. It is essentially a 5-digit mechanical counter equipped with constant-type electrical contacts. Eleven wires are brought out of each decade, one for each unit and one common. These wires lead to the programmer and scanner and

then to the readout equipment. The detent unit, a solenoid-operated wheel with ten spokes, is inserted between the servo motor and digitizer to center the units decade on a contact when command for readout is given.

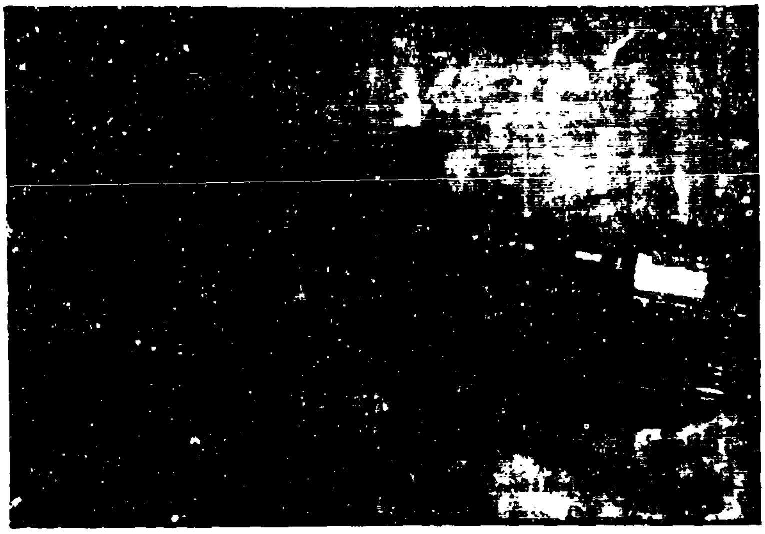


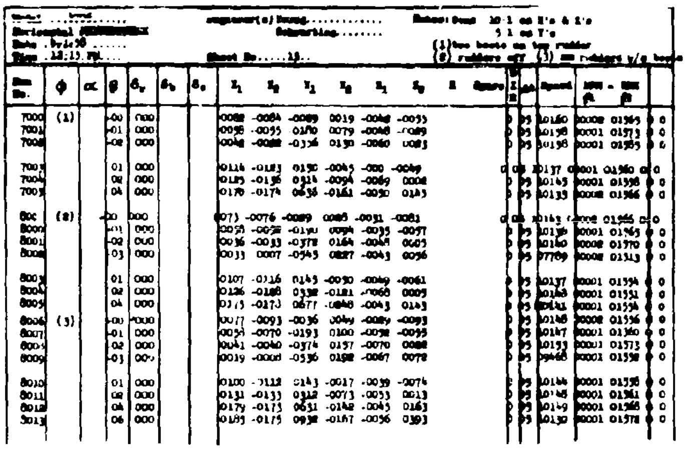
Figure 19 - Digital Indicator

The digital indicator operates in two modes, balancing and readout. In the balancing mode, it is part of the feedback loop, as explained earlier. In the readout mode, the servo motor is automatically stopped and the digitizer serves as a memory which stores the last reading obtained.

The programmer and scanner unit is the brain of the readout system and will be discussed in more detail later. It serves two functions: first, the introduction of predetermined data such as run number, hull angle and control surface angle and secondly, scanning and sequencing the data actively obtained during the test. When a test run is made, the digital indicators are allowed to settle out at an approximately fixed reading. At command, the servo motors are automatically stopped and the scanner unit feeds the readings in correct sequence, one digit at a time, to the solenoid-operated IBM Electric Typewriter. The typewriter tabulates the data on a form especially prepared for the purpose, as shown by the reduced sample given in Figure 20. In addition to the typewritten copy, the data can be transmitted on to punch-cards or tape for processing on high-speed digital computers.

As mentioned earlier, the balance and sensitivity control box contains features that are provided for the purpose of maintaining accuracy and increasing versatility of the system. Among these features are the means of checking zero, adjusting and checking sensitivity, changing

zero reference, and filtering to smooth out the data.



Run No.	ϕ	α	θ	δ_r	δ_b	δ_s	X_1	X_2	Y_1	Y_2	Z_1	Z_2	R	[illegible]	[illegible]	[illegible]	Speed	[illegible]	[illegible]	
7000	(1)		-00	000			0082	-0084	-0089	0019	-0048	-0035			0	05	10160	00008	01565	0
7001			-01	000			0058	-0055	0170	0079	-0048	-0089			0	05	10158	00001	01573	0
7002			-02	000			0048	-0082	-0336	0130	-0060	0083			0	05	10158	00001	01585	0
7003			01	000			0116	-0183	0150	-0045	-000	-0049			0	05	10137	00001	01560	0
7004			02	000			0125	-0136	0314	-0094	-0069	0008			0	05	10165	00001	01598	0
7005			04	000			0170	-0174	0636	-0161	-0050	0143			0	05	10135	00008	01566	0
80c	(2)		-00	000			0073	-0076	-0089	0088	-0031	-0081			0	05	10163	00008	01566	0
8000			-01	000			0053	-0058	-0190	0094	-0035	-0057			0	05	10136	00001	01565	0
8001			-02	000			0036	-0033	-0378	0164	-0045	0005			0	05	10160	00008	01570	0
8002			-03	000			0033	0007	-0545	0227	-0043	0056			0	05	97709	00008	01513	0
8003			01	000			0107	-0116	0145	-0050	-0049	-0061			0	05	10137	00001	01554	0
8004			02	000			0126	-0188	0338	-0121	-0068	0005			0	05	10183	00001	01551	0
8005			04	000			0175	-0173	0677	-0246	-0043	0143			0	05	10141	00001	01554	0
8006	(3)		-00	000			0077	-0093	-0036	0049	-0089	-0093			0	05	10148	00008	01556	0
8007			-01	000			0053	-0070	-0193	0100	-0058	-0055			0	05	10147	00001	01560	0
8008			-02	000			0041	-0040	-0374	0157	-0070	0008			0	05	10153	00001	01573	0
8009			-03	000			0019	-0008	-0536	0198	-0067	0078			0	05	09468	00001	01558	0
8010			01	000			0100	-0112	0143	-0017	-0039	-0074			0	05	10144	00001	01558	0
8011			02	000			0131	-0133	0312	-0073	-0053	0013			0	05	10148	00001	01561	0
8012			04	000			0179	-0173	0631	-0142	-0045	0163			0	05	10149	00001	01568	0
8013			06	000			0185	-0175	0938	-0167	-0056	0393			0	05	10130	00001	01578	0

Figure 20 - Typical Data Sheet for Static Stability and Control Tests

A zero-check switch is provided to separate any change of reading due to causes other than actual gage displacement. These changes could be due to causes such as changes in value of circuit resistors or diodes. When the switch is closed, the primaries of two input transformers of the control circuit are connected in parallel so that their voltages must be equal regardless of gage core position. If the zero-check reading differs from the original value, the difference is due to changes in the control unit circuitry rather than the gage. Thus the reading obtained on the digital indicator may be corrected by this amount. If the gage is not balanced at the time of testing due to preload or core offset, it is desirable to balance it directly. This is accomplished by a "gage zero" potentiometer which is adjusted to make the impedances across the two gage halves equal.

The "pen position" adjustment is provided to set the initial reading of the digital indicator or recorder to any desired value while the model is at rest. The usual practice for steady-state tests is to adjust the digital indicator to read zero when there are no hydrodynamic loads on the system. The setting is periodically checked before each run or group of runs to maintain the zero. The advantages of this procedure are that it provides a means for determining easily whether any changes other than hydrodynamic have occurred in the total system and it eliminates the need for subtracting arbitrary readings on each channel to obtain the net readings. The pen position adjustment is accomplished by a potentiometer which is connected in parallel with the feedback potentiometer.

A span or sensitivity adjustment is provided to establish the calibration of the digital indicator in terms of the load on the gage. The span control is a potentiometer which is placed in series with the part of the circuit that goes with the feedback potentiometer slider. Thus the unbalanced voltage resulting from displacement of the gage appears across the span potentiometer as well as any other resistors placed in series with it. The range of sensitivity varies from nearly zero to an amount somewhat in excess of that required to accommodate the maximum sensitivity of all the types of transducers used in the tests. The span potentiometer has a calibrated index and can be locked into place.

Span control settings are usually established with the modular force gage or roll balance mounted on a calibration stand. Since all components of the measurement system are linear, the settings are determined on the basis of that required to give a reading of exactly 1000 counts on each channel for some predetermined load. As mentioned earlier, the usual sensitivity is 200 pounds for 1000 counts, however, sensitivities of twice or one-half of this amount are used from time to time depending on the range of loads encountered in the test. The sensitivity settings for each calibration are recorded in a log book. These settings have been found to hold true for any particular gage and control box combination over periods as long as a year. The sensitivity control settings are locked into place by the test engineer prior to the test. The settings are checked independently by another engineer to avoid errors.

To give further assurance that nothing has occurred to change sensitivity and to make the control boxes and gages interchangeable during the course of a test, a checking system which is independent of the transducer movement is provided. This "span check" is made by applying a step signal to simulate an actual transducer change. To do this, a precision resistor in the control box is shunted across one gage coil.

The signal coming from the gages is a fluctuating one even in steady-state tests. This is due largely to carriage vibrations which are transmitted to the model through the rigid attachment. A filter is provided in the control box to smooth this signal to obtain one steady value at the digital indicators. The filters are made up in steps so that only the amount needed for smoothing is used without needlessly sacrificing speed of response. The filter switch connects successive values of capacitance between the span potentiometer slider and one side of the feedback potentiometer. The polarity between these two points is always the same so that electrolytic capacitors of reasonable size can be used. Since this capacitance is outside of the servo feedback loop, it introduces no instability.

Recording System for Oscillation Tests

The recording system for the oscillation tests is the same in many respects as that used for steady-state tests. The distinguishing features are: the introduction of the synchronous switch, the introduction of the force-component separator and integrator, and the use of the Brown

Recorder in lieu of servo amplifier and digital indicator. These components are shown by the lower leg of the block diagram of Figure 18. They become part of the recording system when the selector switch is thrown from static to dynamic.

The synchronous switch shown by the block diagram is the one that was described in detail in connection with the forced-motion mechanism. It is, however, more properly a basic component of the recording system for the oscillation tests. The one switch assembly is common to all channels. It is used in conjunction with the separate channels of the force-component separator to resolve the sinusoidal gage signals into in-phase and quadrature components, as shown by Figures 3 and 4.

The force-component separator can be seen in the top of the second rack from the left in Figure 17. It is made up of 8 channels to serve each individual force or roll gage. Each channel is a separate plug-in unit and is interchangeable with all of the others. The major components of the force-component separator unit are the selector switch and polarized micro-second relay.

The selector switch has five settings. The switch is set prior to a given run or group of runs to determine the mode of operation of the recording system. One setting activates the recording system for steady-state tests, as shown by the block diagram. The other four settings are pertinent to the oscillation tests. They are divided into an in-phase pair and a quadrature pair, denoted briefly as "in" and "out", respectively. Each pair is further subdivided into "normal" and "reverse". The "in" setting electrically actuates the O-180-degree pair of synchronous switches; the "out" setting electrically actuates the 90-270-degree pair. A change of setting from normal to reverse, changes the polarity of the signal. One-half the algebraic sum of the integrated normal and reverse signals for any given component is equivalent to the mean direct-current level of the alternating signal. The major advantage of the normal-reverse procedure is that it eliminates the requirement for knowing the static zero precisely.

After the selector switch setting is made, the micro-second relay comes into play with the given synchronous-switch pair to perform a rectification of the signal. For example, if an "in-normal" setting is made, then the O-180-degree pair of synchronous switches is electrically active. Everytime the O-degree switch is thrown, the coil in the relay is energized to close a set of contacts which complete the electrical circuit. The signal then comes through with normal polarity. Everytime the 180-degree switch is thrown the coil in the relay is energized to close a set of contacts which reverse the polarity or "flip" the signal. The net effect of this alternate flipping at O and 180 degrees is a rectified signal which goes to the recorder. Since the synchronous switch has been phased-in with the motion by adjusting both the phase-changer and switch on the forced-motion mechanism, the rectification is in synchronism with the motion.

The integrator consists simply of 8 channels of long-time-constant filters. A filter unit consisting of an assembly of electrolytic capacitors is housed in each of the force-component separator units. A 5-point switch is provided with each filter unit to obtain capacitances of 1000, 2000, 4000, 6000, and 8000 microfarads, respectively. For any particular test, the filter step is chosen on the basis of the minimum amount of capacitance required to give the desired smoothing. In this way, the time required to obtain a complete run is minimized. Since the integration or filtering is done with capacitance changes rather than resistance changes there is no attenuation of the steady-state signal in going from no filter to full filter.

The integrated signal is equivalent to the direct-current level of the oscillatory signal. Consequently, it is necessary to multiply the magnitude of the integrated signal by a factor of $\frac{\pi}{2}$ to arrive at the amplitude of the component being measured.

Although, it is possible to record the integrated signal on the digital indicator, it has been found more advantageous to resort to a Brown Recorder for this purpose. The running record on the strip chart enables the test engineer to judge when the time constant of the filter has been exceeded and a condition of balance has been established. Also, any remaining "ripple" can be averaged by eye to obtain a more refined value.

The validity of the force-component-separator and integrator has been established on basis of controlled laboratory tests. The core of a magnitude unit was oscillated approximately sinusoidally at each of two frequencies and at a known amplitude. Predetermined phase angles were varied over a range of about ± 40 degrees. The results of such a study are shown in Figure 21 which compares measured values with known or precisely computed values. The study has demonstrated that the amplitude of any individual component is determined to an accuracy of within 1 percent. This is comparable with the accuracies usually obtained in refined steady-state tests.

Programming and Control Equipment

To increase versatility and make the system complete, equipment is provided which enables remote control of model and test conditions from the Penthouse. The major advantages of this equipment are:

1. It conserves the number of personnel required for testing.

2. It saves a substantial amount of the time required to make changes in test conditions.

3. It allows the changes to be made while the carriage is underway. For example, in one pass up the basin, 4 or 5 conditions can be evaluated in static tests instead of one, as was the case where manual means were employed to make changes.

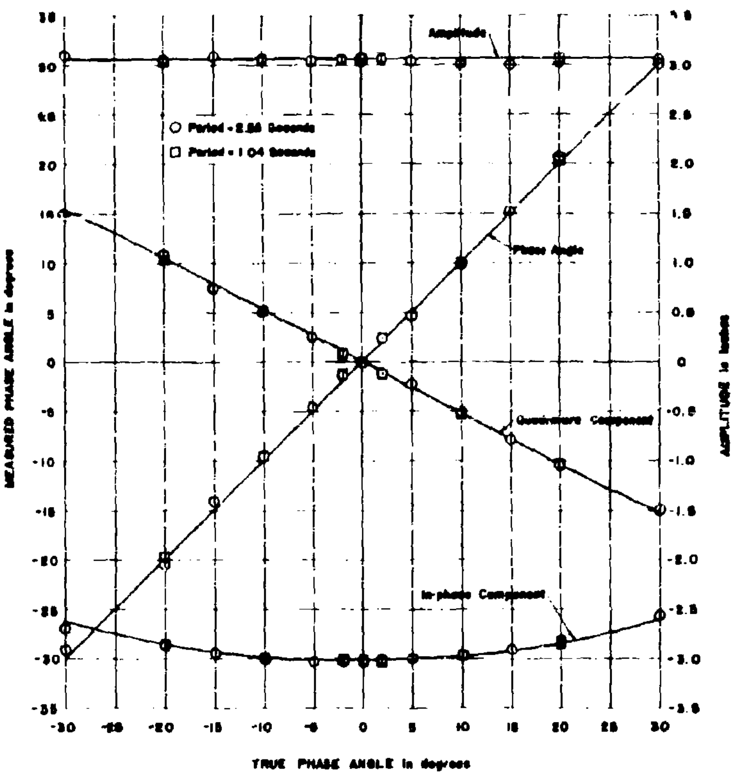


Figure 21 - Evaluation of Force-Component Separator and Integrator System

Most of the programming and control equipment is contained in the first rack on the right shown in Figure 17. The equipment incorporates features for control of hull angles (tilt table settings), control surface angles, propeller rpm, carriage speed, and oscillation frequency.

The tilt table control consists of two switches, one which determines whether the direction of the motion will be up or down and the other which specifies the number of degrees orientation. In this manner, discrete angles in one-degree steps can be obtained over a range of ±20 degrees. When a setting is made on the control panel, the motor-driven screw-jack on the tilting mechanism is actuated and runs until the appropriate micro-switch is reached. Signal lights on the panel indicate when the tilt table has come to rest at the discrete angle. The settings can be made while the carriage is underway. Consequently, several hull-angle conditions can be tested in one pass along the basin.

The control surface angles are set with a system which utilizes a digital indicator of the type used in the recording system. A switch on the panel is used to start, stop, or reverse, the motor-driven rotary actuator units installed in the model. A separate actuator is attached to each set of control surfaces. The output shaft of each actuator has a potentiometer which is used as a transducer to indicate shaft rotation. Prior to testing, the potentiometer is calibrated with the digital indi-

cator to determine the degree settings. A control surface angle is set by starting the actuator motor, watching the digital indicator until the desired angle is reached, and then stopping the motor. The control-surface angles can be changed underway to obtain a number of discrete angles in one pass along the basin.

The system for setting propeller rpm is somewhat more elaborate. The speed of the propeller motors in the model is governed by an amplidyne control system which operates with a 0 to 400 volt, 5-kilowatt motor-generator set installed on the carriage. The speed of the propeller motors is varied by changing the voltage coming from the generator. This is accomplished by a control box with two Heliopots which are placed in the generator field circuit. One of the heliopots is used either completely open or completely closed. The other is used to obtain fine adjustment. In this way, power can be gradually applied with the first Heliopot and yet the fine adjustment on the other can be retained to maintain constant propeller rpm from one run to another.

The propeller rpm is indicated on a revolution-speed-time recorder which is made up of Berkeley-type electronic pulse-counters. The revolution pulses are provided by a pick-up installed within the model on the propeller shaft. The pulses are counted over a finite time interval of 1, 5, 6, or 10 seconds to give a direct display of rps or rpm. The time interval is accurately determined with a millisecond timing oscillator.

The propeller rpm for a given test condition is predetermined by one or a combination of three methods: inspection of a frequency meter which operates off the pick-up units, inspection of the counts indicated on the revolution-speed-time recorder for 1-second time intervals, or inspection of readings of the X-gages on the digital indicators to determine whether the correct propeller thrust is being delivered. When a model has more than one propeller, the shafts are usually geared together so that a pick-up is used only on one shaft.

Remote control of the carriage speed is not required since it is hydraulically driven and speed is maintained within 0.01 knot once the setting is made. The speed settings are provided to the carriage operator by the test engineer using an inter-communication system. The carriage speed is indicated on the revolution-speed-time recorder in much the same way as the propeller revolutions. An electromagnetic pulse generator on an idler wheel of the towing carriage produces 100 pulses per foot of travel of the carriage. The number of pulses taken over the same finite intervals as the propeller revolutions gives a direct display of speed in feet per second or feet per minute.

The oscillation frequency is governed by a constant speed induction motor. Consequently, only an off-on switch is required on the control panel. The setting on the pulley drive determines which one of the three discrete frequencies will be imparted to the drive-shaft. A millisecond timer operating off a micro-switch on the drive-shaft of the mechanism

has been employed to check and record oscillation frequency. It has been observed, however, that there has been no measurable change in frequency for each of the three settings since the system was first installed.

The values of the finalized settings made with the programming and control equipment pass into the scanner unit shown by Figure 18. The values are thus sequenced and recorded automatically and appear on the data sheet as shown in Figure 20.

TYPICAL TEST RESULTS OBTAINED WITH SYSTEM

All of the hydrodynamic coefficients required in the equations of motion for submerged bodies in six degrees of freedom can be obtained with the DTMB Planar-Motion-Mechanism System. This is accomplished by appropriate orientation of the model and mode of operation of the system. With present techniques, a few of these coefficients are determined only over their linear range. It is believed, however, that when the capabilities of the system are fully exploited, techniques will evolve for extending these few coefficients into the nonlinear range, if found to be necessary.

It is beyond the scope of this paper to discuss all of the numerous coefficients which enter into the equations of motion. It is believed to be pertinent, however, to include representative samples of test results obtained for each of the three classes of coefficients. Examination of these samples should provide insight not only into the nature of these coefficients but also the quality with which they can be determined by the system. Before proceeding, it is reemphasized that all of the coefficients are obtained from explicit relationships.

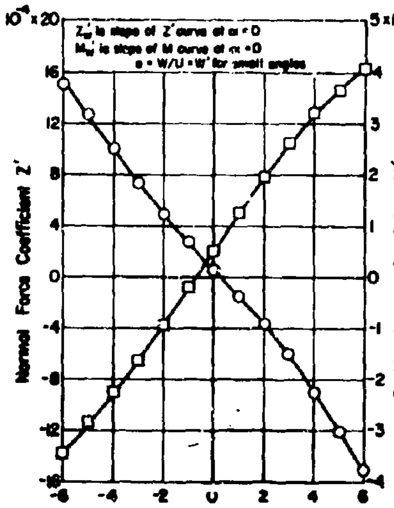
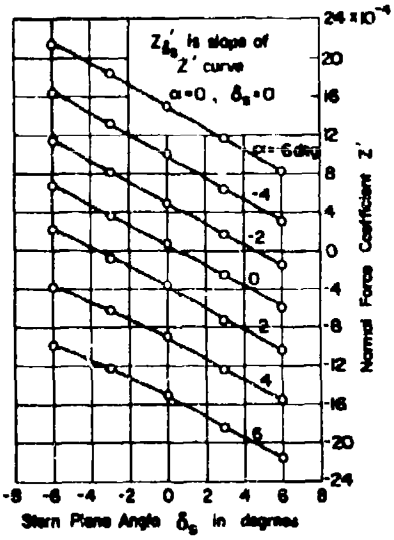


Figure 22 - Typical Curves of Static Force and Moment versus Body Angle

Typical test results for coefficients of the "static" variety are shown in Figures 22, 23, and 24. The variation of normal force and pitching moment with body angle is shown in Figures 22 and the variation of normal force and pitching moment with stern plane angle is shown for various body angles in Figures 23, and 24, respectively.



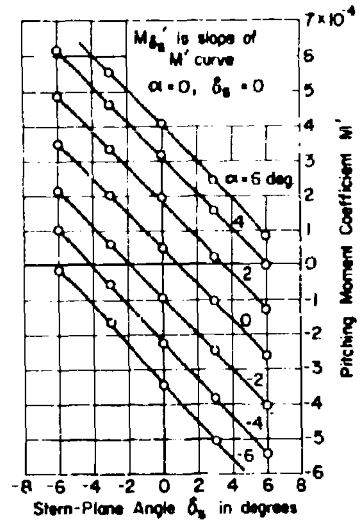


Figure 23- Typical Curves of Static Force versus Control-Surface Angle for Various Body Angles

Figure 24- Typical Curves of Static Moment versus Control-Surface Angle for Various Body Angles

As indicated in the legends of the figures, the derivatives are obtained from the slopes of the appropriate curves faired through the data points. The slopes of the body-angle curves are taken through a body angle of zero and become the static stability derivatives. The slopes of the control-surface curves for zero body angle are taken through a control-surface angle of zero and become the control derivatives.

Typical results of pure pitching tests which are used to obtain (damping) force and moment derivatives are shown in Figure 25. It may be noted that the quadrature components of force measured at each of the two struts are plotted separately. It has been found desirable to do so since the slopes of the two curves can then be substituted directly into the two formulas shown in the legend to obtain the damping force derivative Z_q and damping moment derivative M_q.

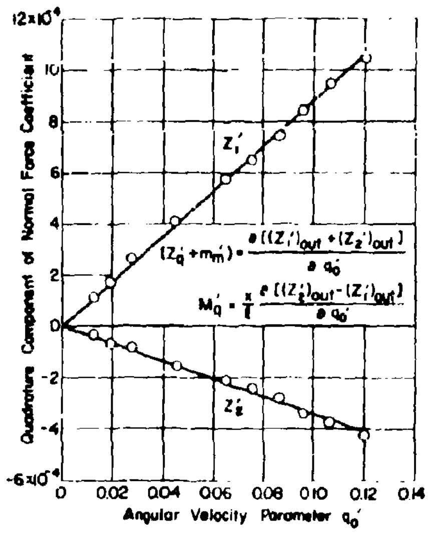


Figure 25 - Typical Curves of Forces versus Angular Velocity Amplitude from Pure Pitching Tests used to Obtain Damping Force and Damping Moment Derivatives

The curves of in-phase force components shown in Figure 26 are also typical of the results obtained from pure pitching tests. Here again, the force components measured at each strut are plotted separately. The angular acceleration derivatives, the added moment of inertia $M_{\dot{q}}$ and associated force $Z_{\dot{q}}$, are obtained from the slopes of the curves using the formulas given in the legend.

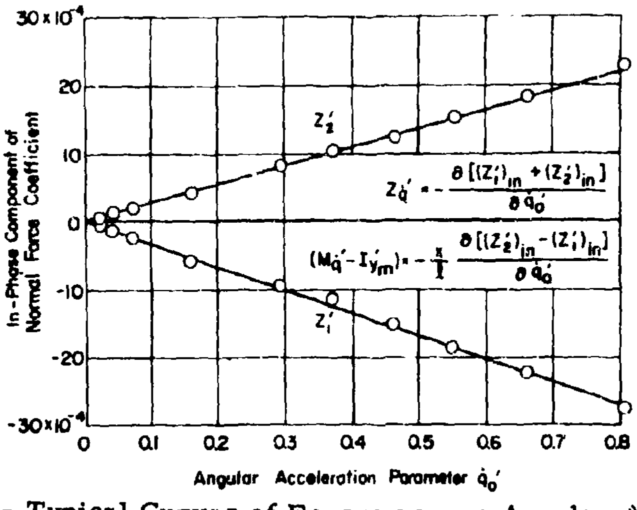


Figure 26 - Typical Curves of Forces versus Angular Acceleration Amplitude from Pure Pitching Tests used to Obtain Added Moment of Inertia (and Associated Force)

The kind of results obtained from pure heaving tests is shown in Figure 27. The slopes of the separate in-phase force component curves are used with the formulas given to obtain the linear acceleration derivatives, the added mass $Z_{\dot{w}}$ and associated moment $M_{\dot{w}}$.

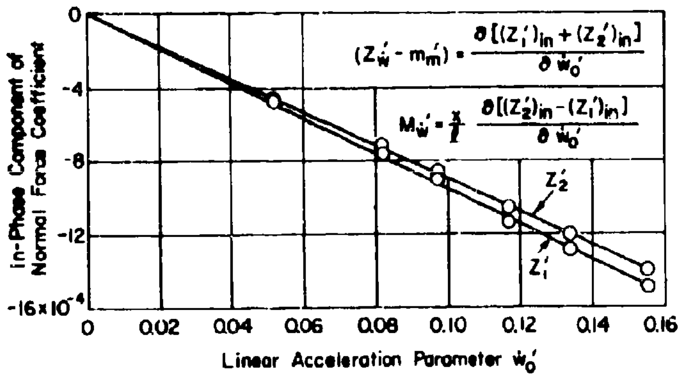


Figure 27 - Typical Curves of Forces versus Linear Acceleration Amplitude from Pure Heaving Tests used to Obtain Added Mass (and Associated Moment)

ACKNOWLEDGMENT

The DTMB Planar-Motion-Mechanism System was conceived and developed jointly by the Author and Mr. Alex Goodman, both members of the staff of the Hydromechanics Laboratory of the David Taylor Model Basin. Patent proceedings have been initiated in behalf of the United States Navy Department with the names of Messers. Gertler and Goodman as originators of the system. The originators wish to express their gratitude to the many members of the Industrial Department of the Model Basin whose contributions and efforts in the design and construction of components made the ultimate system possible. Particular thanks are due to Messers. M. W. Wilson, J. E. Stern, T. G. Singleton, G. J. Norman, J. W. Day, P. P. Day, C. W. Scott, J. G. Tisdale, R. G. Hellyer, and E. J. Mosher, all of the Industrial Department.

Because this report was previously distributed in a different form, no Initial Distribution List is necessary.

GPO 927-643